

HEAT BALANCE OF A HVID ENGINE

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ARMOUR INSTITUTE OF TECHNOLOGY

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HEAT BALANCE OF A HVID ENGINE

A THESIS

PRESENTED BY

L. E. JONES AND E. C. SCHWACHTGEN

TO THE

PRESIDENT AND FACULTY

OF

ARMOUR INSTITUTE OF TECHNOLOGY

FOR THE DEGREE OF

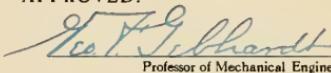
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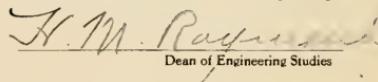
IN

MECHANICAL ENGINEERING

MAY 27, 1920

APPROVED:


H. T. Gilhardt
Professor of Mechanical Engineering


H. M. Page
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The Heat Balance of a Hvid Engine

HEAT BALANCE OF A HVID ENGINE

AUTHOR'S NOTE

It is not our purpose to offer a complete analysis of the Hvid engine. Rather, the work we have done is merely a beginning, and our time has been largely spent in putting the engine in condition to test. This work is described in detail in the text.

We wish to thank Professor Roesch for the opportunity he has given us to work on our own initiative, and for his constant aid and advice.

Professor Perry has also lent us his aid in the shaking-forces analysis.

We are further indebted to Mr. Larson of the class of 1917, who has done some special work on the characteristics of different fuels in the injector. This work will be presented by Mr. Larson personally.

L. E. Jones

E. C. Schwachtgen

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Heat Balance of a Hvid Engine

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The vivid Engine

HEAT BALANCE OF A HVID ENGINE

THE HVID ENGINE

The Hvid Engine is of necessity a heavily built machine and is designed primarily for heavy duty work. It is built in single and multi-cylinder models, and is used largely for farm and machine shop work. A four cylinder 25 horse-power engine is built by the Commonwealth Motors Co. of Joliet, Ill., for marine work. The price of this engine (January 1920) is \$2000. This model should prove popular, despite the high first cost (which by the way, is as cheap or cheaper than any other successful kerosene engine adapted to this work) because of its reliability and inherent economy.

The engine used (Fig. 1) is built by Sears Roebuck & Co. of Chicago, under the trade name "Thermoil". It is of the single cylinder horizontal type and develops 8 horse power. The price is \$300. Early models of this engine gave trouble, due to incomplete development of

Cross Section of Cylinder and Cylinder Head to Show Operation of Engine

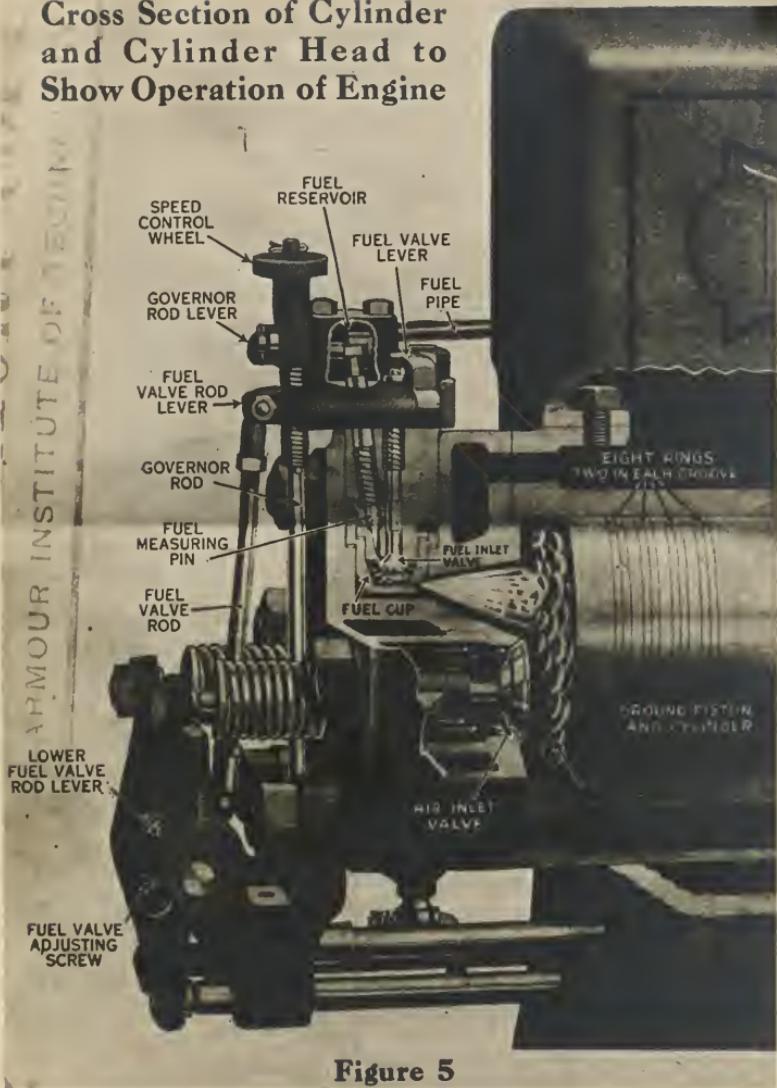


Figure 5

the injector, but this obstacle has been overcome and the present models are considered as reliable as the ordinary farm engine.

The engine is governed by a high speed centrifugal governor, geared to the timing gears. Governing is effected by reduction of fuel, and a full-load speed of 450 r.p.m. is maintained. (Hvid engines have been built that operate successfully at 1500 r.p.m.).

The inherent advantages of this type of engine are:

1. Mechanical Simplicity
2. Low fuel consumption at all loads
3. Absence of electrical or hot bulb ignition
4. Ability to start cold on any free-flowing fuel
5. Absence of carburetting device
6. Low water jacket losses
7. Ease of lubrication
8. Lack of carbon troubles
9. Constant compression
10. Good torque characteristics

EXPLANATION OF CYCLE

The basic four cycle (Otto cycle) principle is used, with this modification, pure air, instead of a mixture of air and gaseous fuel, is drawn in the cylinder on the suction stroke. Two valves are employed, actuated as in the usual gasolene engine by a camshaft geared from the crankshaft to run at one-half engine speed.

Ignition is effected by the compression temperature (900 to 1000 deg. fahr. due to a compression pressure of 425 to 475 lbs. per sq. in.). Referring to figure 2, the phases of the cycle are as follows:

SUCTION STROKE: As the piston moves down the stroke (toward the crankshaft), the inlet valve "A" opens and a charge of pure air is drawn into the cylinder. The fuel valve "B", (actuated by the push rod "R") opens simultaneously with the inlet valve, and fuel enters the cup "C", due partly to the action of gravity and partly to inhalation. The amount of fuel admitted is regulated by the metering -

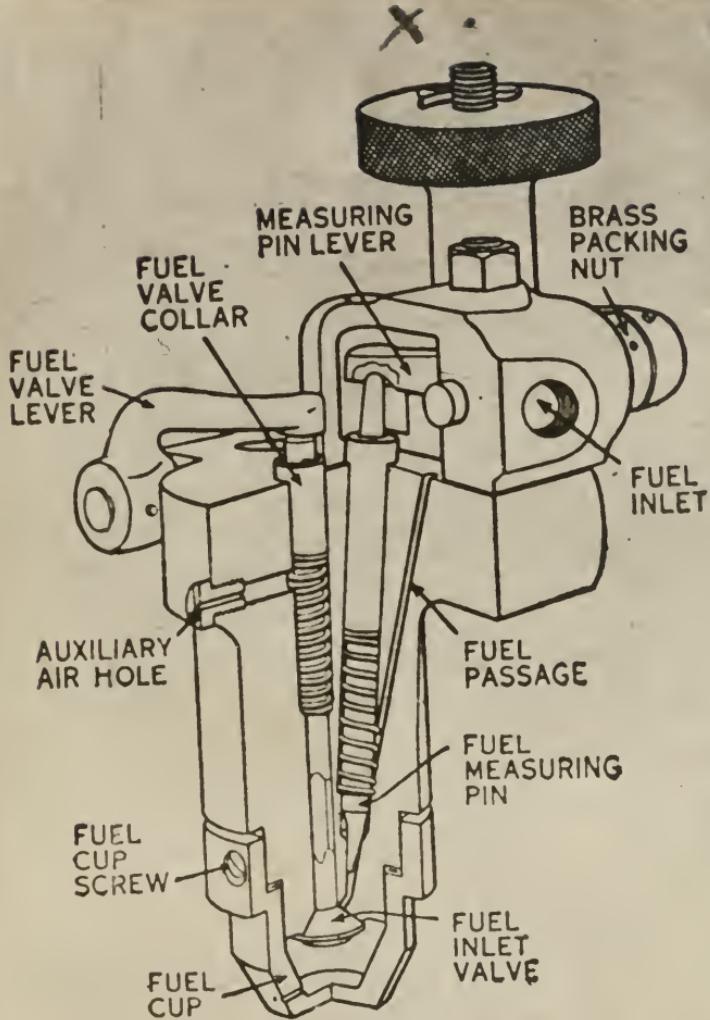


Figure 6.

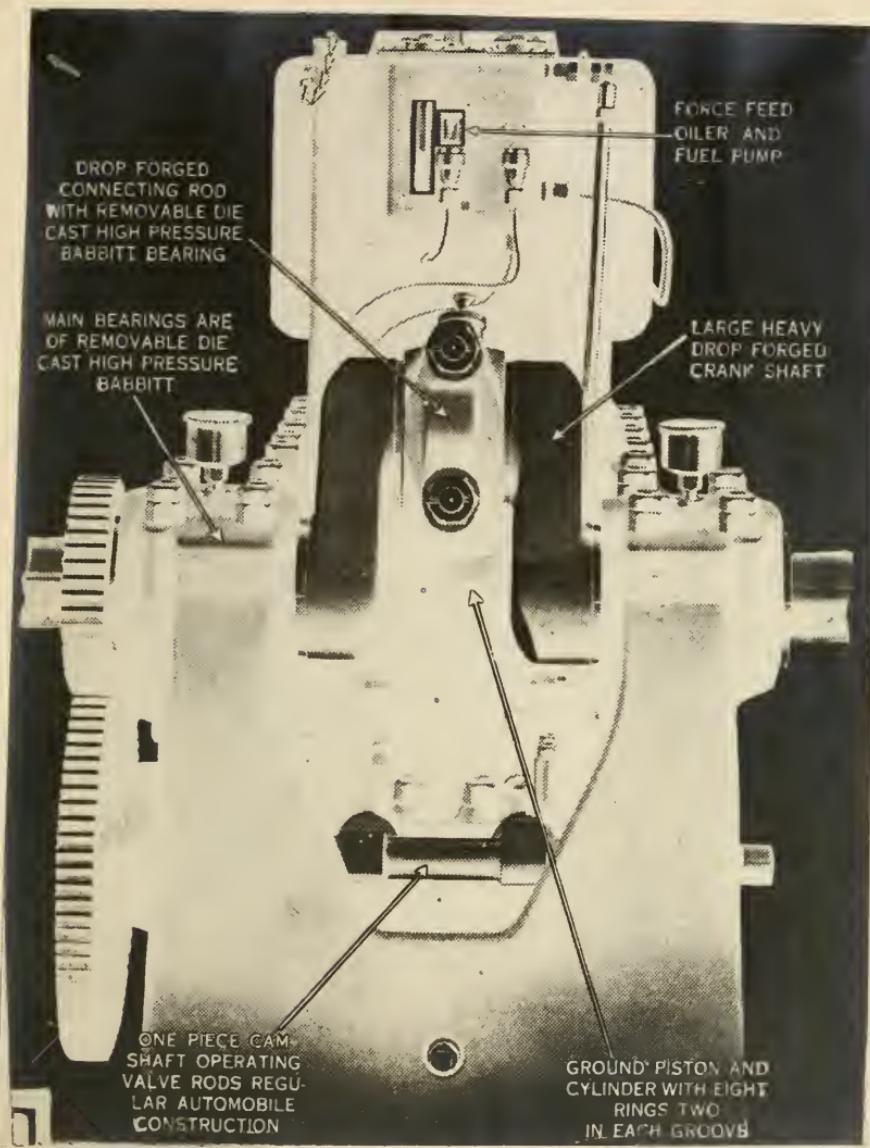
pin "E", which is linked to the governor. As the fuel enters the cup, a small charge of air is drawn through the hole "F" and enters the cup through the guide "G". As the piston nears the end of the stroke, the inlet and fuel valves are closed.

COMPRESSION STROKE: On this stroke all the valves are closed and the entrapped air is compressed to a pressure of 425 to 475 lbs. per sq. in. As the pressure builds up, air is forced into the cup through the holes "H" and a pressure nearly equal to that in the cylinder is finally attained. At the temperature due to this pressure, the more volatile parts of the fuel ignite spontaneously, and a "primary" explosion takes place within the cup. The force of this explosion drives the heavier constituents of the fuel thru the holes into the cylinder. At this stage the piston has reached the end of the stroke and the compression is at a maximum, and the temperature is high enough to ignite the fuel. (The fuel

burns, rather than explodes). Due to air deficiency, the amount of fuel burned in the cup is infinitesimal.

POWER STROKE: As the fuel is forced into the cylinder it is thoroughly atomized, and is ignited by the temperature due to compression. The consequent expansion forces out the piston, as in the ordinary gas engine.

EXHAUST STROKE: As in the ordinary four cycle engine, the exhaust valve opens at the end of the power stroke and the products of combustion are expelled by the piston on the return stroke.



OPERATION

SPEED CONTROL: The speed control wheel (Fig. 1) regulates the amount the fuel valve opens, and thus regulates the amount of fuel fed per stroke to the fuel cup. When the wheel is screwed up as far as it will go, the maximum valve opening is allowed, and the engine runs at normal speed. Screwing the wheel down reduces the supply of fuel and hence reduces the speed.

A noticeable amount of play is observed when the wheel is screwed up. If the wheel is screwed down until this play is eliminated, the fuel supply is entirely shut off. (In order to cut off the supply of fuel quickly, in case of an accident, we inserted a stop-cock in the fuel line, close to the injector).

LUBRICATOR AND FUEL PUMP: The pressed steel box attached to the back of the water hopper (Fig. 3) contains a combination lubricator and fuel pump. The box is divided into two sections, one an oil reservoir and one a fuel reservoir. Oil is pumped from the oil reservoir to the pis-

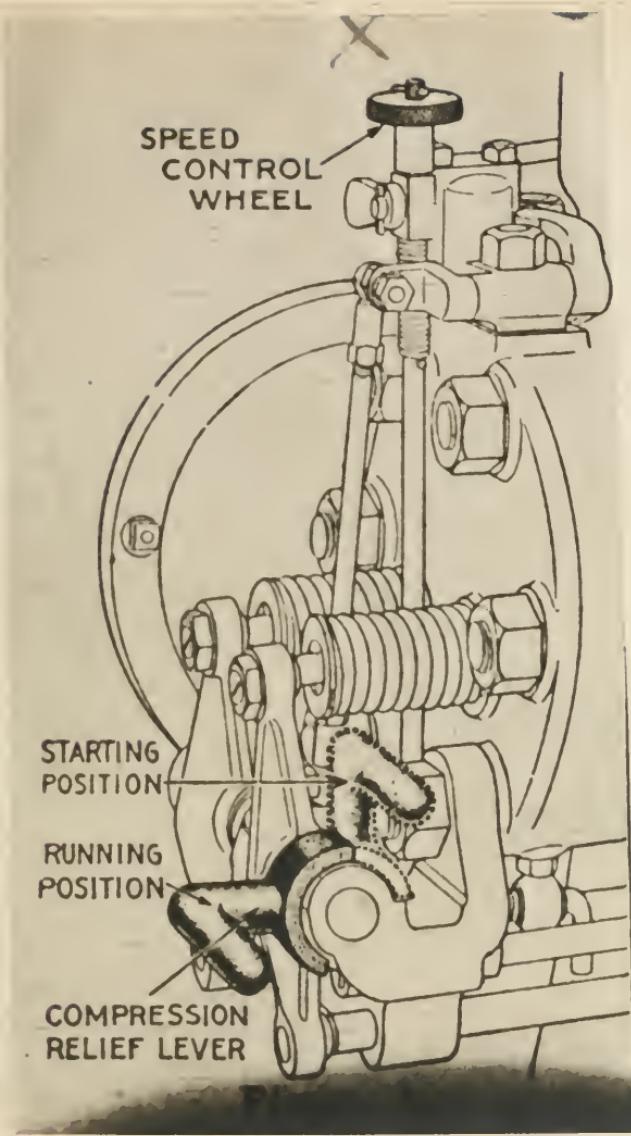
ton, and fuel is pumped from a tank in the base to the fuel reservoir. (The oil reservoir is filled by hand). From the fuel reservoir, the fuel flows under the action of gravity to the injector. The pumps are actuated by the cam-shaft and the link, as shown in Fig. 3.

TO START THE ENGINE: Before starting the engine, the oil and fuel reservoirs must be filled, and all grease cups should be screwed down. To quote from the instruction book furnished by the manufacturers:

"Release the compression in the cylinder by pulling the compression release lever up to the starting position (Fig. 4).

Turn on fuel by screwing up the speed control wheel.

With the starting crank, turn the flywheels rapidly through eight or ten revolutions. Remove the crank rapidly and immediately push the compression release lever down to the running position. The momentum of the flywheels will turn



the engine over compression and the engine will start.

To shut down the engine, screw down the speed control wheel until the fuel is shut off."

The lubricator should be adjusted to give about 15 drops of oil per minute.

If the engine is to be started warm (i.e. after it has been in operation, and has not cooled off) less fuel will be required. Therefore the speed control wheel should be only partially screwed up. Indeed, it is possible to "flood" the engine by admitting too much fuel, in which case difficulty will be encountered in starting.

If the engine pounds upon starting, an excess of fuel is indicated, and the entire supply should be cut off until this excess is used. When the engine begins to slow down, the wheel should be screwed up until proper running conditions are reached.

FUEL INLET VALVE: The fuel inlet valve sometimes leaks, due to the formation of carbon, or

to dirt. When this condition occurs, the valve must be ground in. To quote further from the instruction booklet:

"To grind the fuel inlet valve, remove the fuel injector from the cylinder head. Take off the fuel cup and remove valve collar and valve. Place a minute quantity of powdered pumice and oil on the valve seat, and turn the valve slowly back and forth on the seat. A gentle pressure should be used. After grinding, the parts must be washed in kerosene."

In placing the indicator on the cylinder head, the rods which actuated the fuel valve had to be replaced with special rods, made to clear the indicator. As these rods were of different length from the original ones, the timing of the valve was changed. Resetting was accomplished with the crank straight down (180 degrees from the position shown in Fig. 3) and the compression release lever in the running position. At this phase, there is supposed to be between .005 and

.01 of an inch clearance between the valve tappet and valve stem. This clearance is adjusted by the fuel valve adjusting screw shown in Fig. 1.

Work Preliminary to Testing

WORK PRELIMINARY TO TESTING

MOUNTING THE ENGINE: We received the engine bolted on skids. These were left in place and a bolt was sunk in the concrete floor at each end of either skid. (Special bolts, having a nut on each end, were used. Holes were bored in the skids first, and then the relative positions of these holes were laid off on the concrete floor. Holes were drilled with a star drill and a nut with a bolt threaded in place was inserted in each hole. Melted lead was then poured around each bolt. When the engine is removed, the bolts can be readily withdrawn from the nuts, leaving them in the floor for future use.

The engine was run unloaded after being bolted in place, but excessive vibration resulted. Accordingly, two more holes were placed in each skid, and bolts to correspond were sunk in the floor. A second test proved the vibration to be satisfactorily reduced.

For a permanent setting the engine should be

mounted directly on a concrete base.

THE REDUCING MOTION

To indicate the engine a special head with a hole tapped in it to receive the indicator was procured from the manufacturers. The reducing motion, however, had to be made in the shop.

A reducing motion must reproduce, point by point, the motion of the piston. When the engine has a cross-head, the familiar pantograph motion is usually employed, and the length of card produced is about three inches.

As this engine has a full-trunk type piston (customary in gasoline and small gas engine practice) this method could not be used. Our apparatus consisted of a small eccentric, four and one fourth inches in diameter, having a throw of five eights of an inch. This gave an indicator card one and one fourth inches in length. The parts were made small, in an attempt to eliminate inertia forces and the resulting discrepancies.

The eccentric circle was keyed to the crank-shaft at zero degrees with the crank. A push rod extended along the center line of the engine, close to the indicator. Three bearings supported the rod, which was made of steel, three quarters by three eighths of an inch. To reduce weight, the rod was melted out to an I beam section, except at the bearings.

The rod must be placed precisely on the center line of the engine. If it is displaced either way, the motion of the piston will not be accurately reproduced, due to a difference in angularity on the forward and return strokes. (This fact is made use of in shaper practice, where a "quick return" motion is realized by the use of an off-set rod).

The eccentric was placed on the right hand side of the engine, inside the flywheel, and set the flywheel out about seven-eighths of an inch. This seriously interferred with cranking the engine, as room enough was not left to fit

the crank handle properly on the shaft.

Unfortunately, this objection was removed by the failure of the apparatus. After ten minutes of running, the eccentric strap failed, wrecking the entire reducing motion. Whether the failure was due to a seized bearing, or an inherent weakness in the strap, has not been determined. However, it is probable that the strap gave away, as the inertia forces at 450 r.p.m. are large.

In redesigning the apparatus, more rugged construction was used. (See blue print of eccentric appended). The throw was increased to one inch, giving a two inch card. To obviate the difficulty in cranking, the new eccentric was placed on the other side of the engine.

The push-rod used was made of five-eighths cold rolled steel shaft of circular cross-section. Two bearings were used, in place of three, as the larger rod was less subject to flexure. We found by experience that the

simplest way of lining up such a shaft was by babbetting the bearings with the shaft lined up in place. (Previous to the babbetting, a steel bearing was used. This bearing seized, but due to the rugged construction employed, little damage was done, beyond bending the rod).

CONSTRUCTION OF PRONY BRAKE

A two-foot brake drum was made from plans furnished by the Institute. The machine work was done under Mr. Agle's supervision (by the author's).

A large pulley runs "sweeter" than a small one, due probably to the increased rim velocity. It was for this reason that a two foot pulley was used.

A brake band of steel, lined with wood blocks, was made as per the submitted blue print. The brake arm (distance from the center of the crankshaft to the point of application of the load) was made 31.5 inches. Substituting this value in the formula

$$H.P. = \frac{2\pi R N W}{33000}$$

there results the equation

$$H.P. = \frac{N W}{2000}$$

where $N = R.P.M.$

$W =$ net load on scale pan in lbs.

The Heat Balance

THE HEAT BALANCE

We did not have sufficient time to run a complete heat balance, but the engine has been left in condition to run this test.

Several runs for horse power were made, and a typical indicator card (Fig. 5) is included in this report.

At Professor Roesch's suggestion, we have included a heat balance, using data taken from an article in the January 1920 number of the Scientific American Monthly. This article was written by Mr. E. B. Blakely, who used data collected from a series of tests run by Prof. Roesch at the Institute.

Further information can be obtained by referring to the "Proceedings of the American Society of Mechanical Engineers" for December, 1919.

The heat balance gives the mechanical and thermal efficiencies for both brake and indicated horse power. The engine used by Prof. Roesch was the same size and model as the engine we

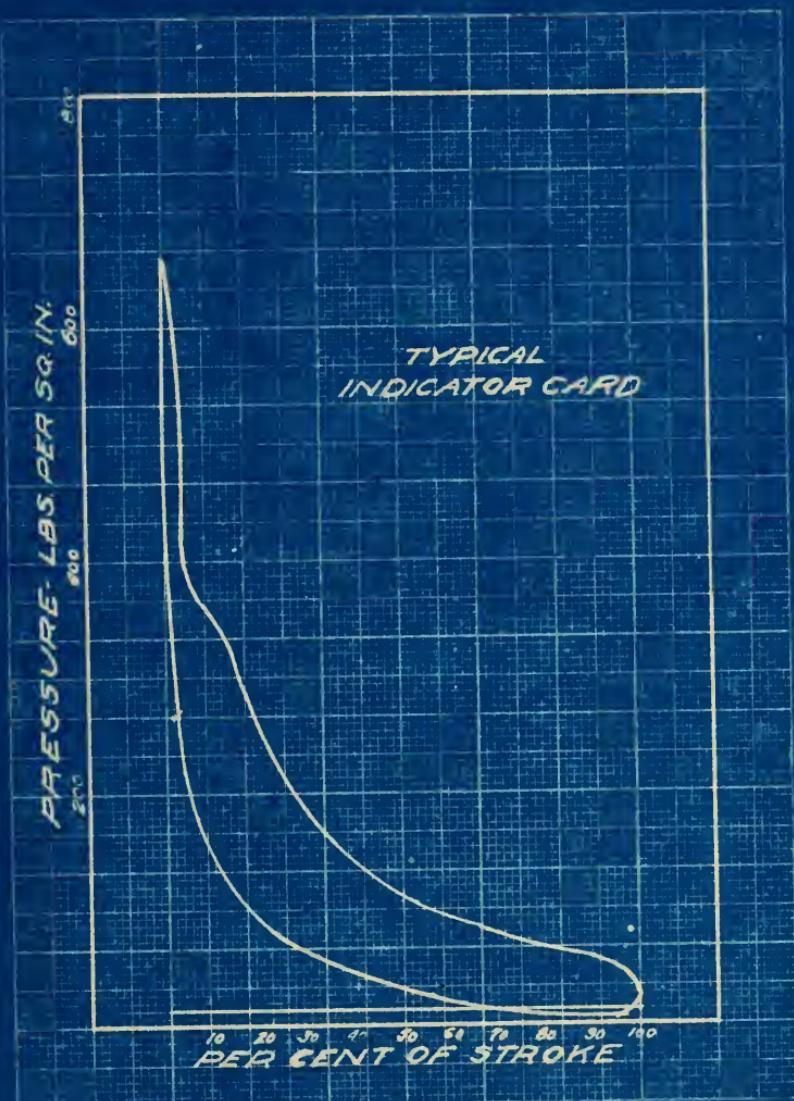


FIG. 7.

used (5-3/4 in. bore and 9-in. stroke).

For testing purposes, the engine was flexibly connected to a Sprague electric dynamometer by two Spicer universal joints.

Engine speeds and fuel weights were obtained by electrically operated devices. The water jacket losses were determined by weighing the water used and recording the temperature differences. The sensible heat of the exhaust gases was determined by the calorimeter method.

Prof. Roesch also took indicator cards, but (to quote from Mr. Blakely's article)" ... because of the probable errors due to the high pressures involved, and the comparatively high speed of the engine, these cards were used merely to study the valve settings and general events of the cycle."

The engine was operated under various loads and speeds with various adjustments of fuel supply, compression and cup design. The final compression pressure was set at 390 pounds per

square inch.

Figures 6 and 7 illustrate by curves the results of these tests.

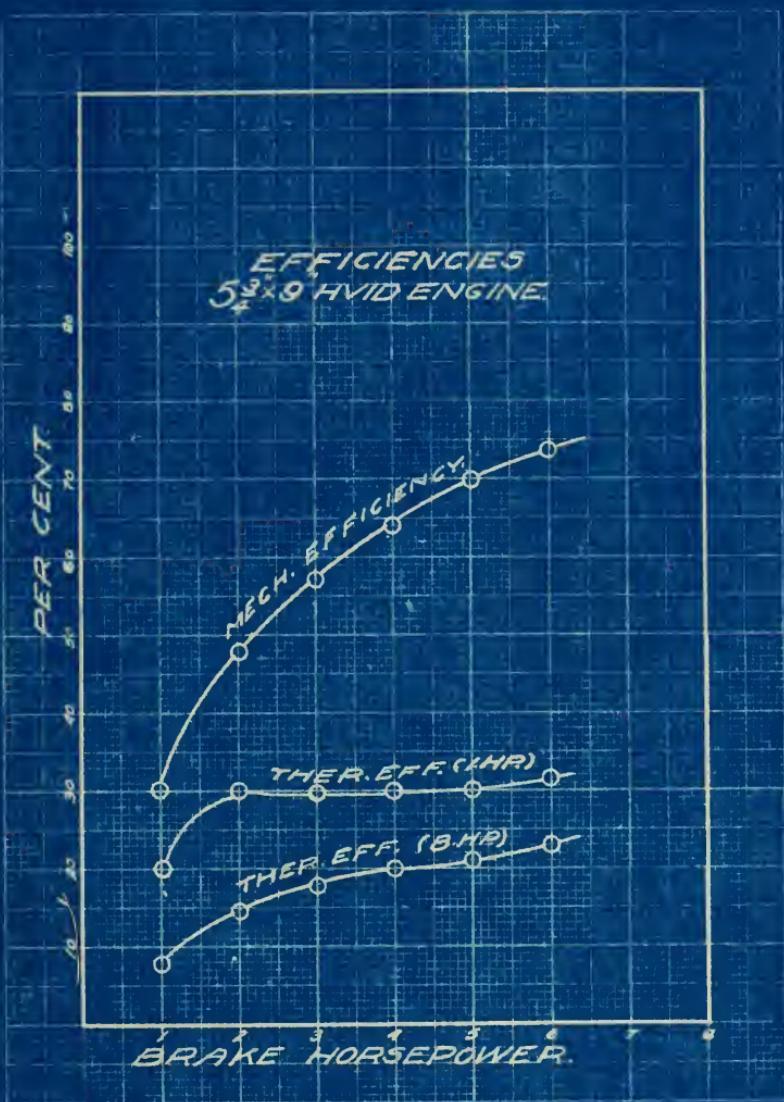


FIG. 5.

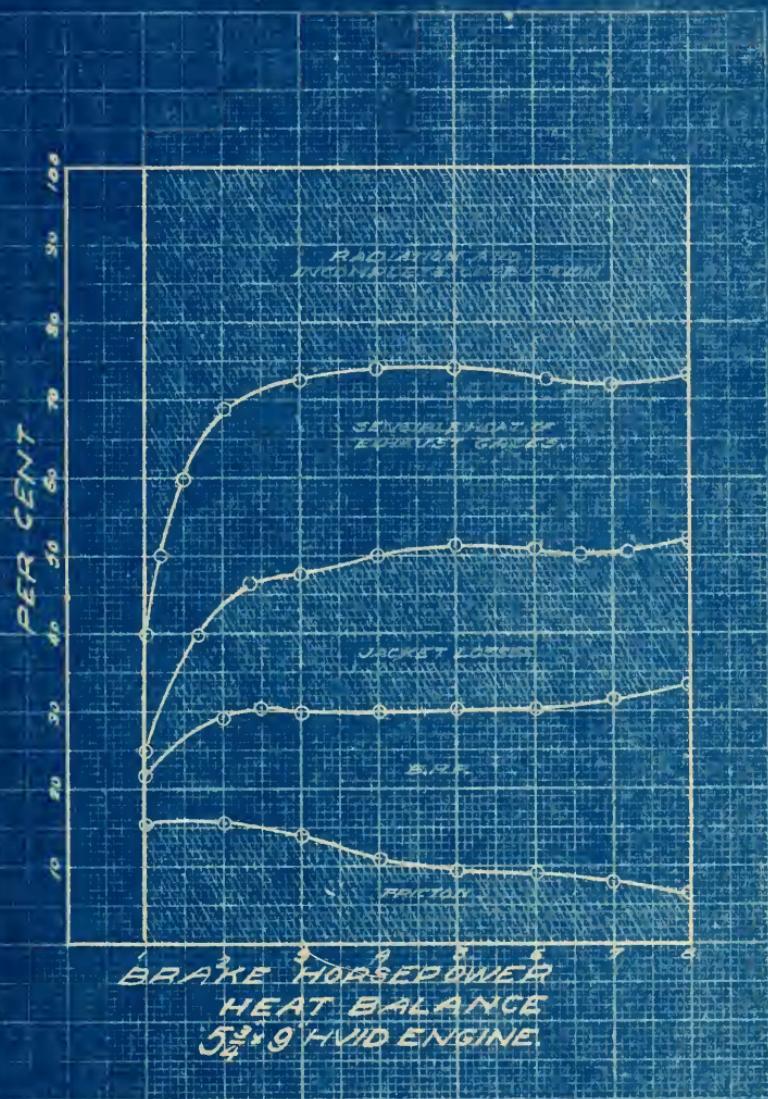


FIG. 6.

Shaking Force Analysis

DETERMINATION OF THE SHAKING FORCES

In addition to the heat balance proper a study of the shaking forces produced in the Hvid engine was made. Blue prints marked plates #1 and #2 illustrate diagrammatically the forces found and calculated in the following pages.

The pressure produced upon ignition of the kerosene vapor is expended in two ways. First, it accelerates the moving parts; second, it produces a pressure on the crank pin which may be resolved into two components, one of which produces rotation of the crank shaft, and the other pressure on the main bearings.

In order to get a clear idea of just how these forces act it is necessary to know something about the relative motion of the various moving parts. The piston has a simple motion of reciprocation, the crank an approximately uniform motion of rotation, while the connecting rod has a complex motion of combined oscillation and reciprocation. The reciprocating motion requires accelerating forces acting

alternately in opposite directions, as does also oscillation. On the other hand, a rotary motion induces a centrifugal force acting always toward the crank pin.

The various forces influencing rotation are illustrated in Fig. A, page where

S = Net gas pressure

F = Translational inertia force of piston and parts

T = Translational inertia force of connecting rod

A = Angular inertia force of connecting rod

C = Centrifugal force of connecting rod

G = Gravity force due to the weight of the connecting rod

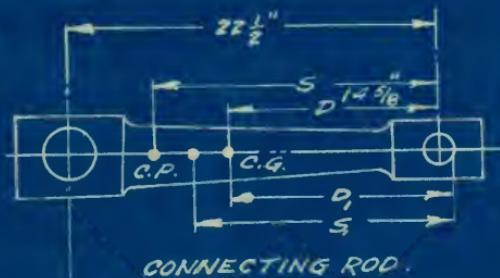
NET PRESSURE

The indicator card shown in plate 1 is the net pressure card, as the engine is single acting. The engine exhausts at $1-1/2$ pounds above atmospheric pressure and the charge is drawn in at $1-1/2$ pounds below atmospheric pressure.

FIG. A.



DIAGRAM OF FORCES



CONNECTING ROD.

TRANSLATIONAL INERTIA FORCE OF PISTON AND PIN

At the end of each stroke the velocity of the piston and pin is zero, while near the middle of the stroke it is a maximum. To speed up these parts from rest to a maximum velocity and then to bring them back to rest again requires an alternately accelerating and retarding force.

This force F is found and calculated as follows:

$$\text{Stroke} = 9"$$

$$r = 1/2 \text{ stroke} = 9/2 = 4.5"$$

$$\text{Piston diameter} = 5.75"$$

$$\text{Piston area} = 1/4 \times 5.75^2 = 20.959 \text{ sq.in.}$$

$$\text{Length of connecting rod} = 22.5"$$

$$\text{Ratio } r/L = \frac{4.5}{22.5} = 1/5$$

$$\text{Weight of piston + pin} = 25 \text{ pounds}$$

$$\text{Mass of piston + pin} = \frac{25}{32.2} = .777$$

$$N = \text{revolutions per sec.} = \frac{450 \text{ r.p.m.}}{60} = 7.5$$

$$F = -M \omega^2 r (\cos. \theta + \frac{r}{L} \cos. \theta)$$

$$-M \omega^2 r = -\frac{M \omega^2}{2} r = -\frac{M \omega^2}{r} = -M(2\pi r)^2 n^2$$

$$- M \cdot 2 n^2 \cdot r$$

$$- 4 \cdot 2 = 39.478.$$

$$M \cdot N^2 \cdot R = \frac{.777 \times 56.25 \times .375}{25.959} = .632$$

$$F = - 39.478 \times .632 \times (\cos. \theta + 1/5 \cos. \theta)$$

DETERMINATION OF $(\cos. \theta + 1/5 \cos. \theta)$

Angle	Function	Numerical Value	
0°	$(\cos. 0^\circ + 1/5 \cos. 0^\circ)$	$(1 + 1/5 \times 1)$	+1.2
30°	$(\cos. 30^\circ + 1/5 \cos. 60^\circ)$	$(.866 + 1/5 \times .5)$	+ .966
60°	$(\cos. 60^\circ + 1/5 \cos. 120^\circ)$	$(.5 - 1/5 \times .5)$	+ .400
90°	$(\cos. 90^\circ + 1/5 \cos. 180^\circ)$	$(0 - 1/5 \times 1)$	- .200
120°	$(\cos. 120^\circ + 1/5 \cos. 240^\circ)$	$(-.5 - 1/5 \times .5)$	- .600
150°	$(\cos. 150^\circ + 1/5 \cos. 300^\circ)$	$(-.855 + 1/5 \times .5)$	- .766
180°	$(\cos. 180^\circ + 1/5 \cos. 360^\circ)$	$(-1 + .2)$	- .800

$$F = - 24.98 (\cos. \theta + 1/5 \cos. \theta)$$

Angle	$(\cos. \theta + 1/5 \cos. \theta)$	Constant	F
0°	+ 1.2	- 24.98	- 29.60
30°	+ .966	"	- 24.05
60°	+ .400	"	- 9.97
90°	- .200	"	+ 4.98
120°	- .600	"	+ 14.98
150°	- .766	"	+ 19.13
180°	- .800	"	+ 19.95

TRANSLATIONAL INERTIA FORCE OF THE CONNECTING ROD

To produce the translation of the rod the

force T Fig. A, page applied at the center of gravity is necessary. This force "T" is resolved into two forces, one "T₁" acting at the pin and the other "T₂" acting at the crank pin.

$$T = T_1 + T_2$$

$$T_1 = - M_C \frac{(L - Z_C)}{L} W^2 N \left(\cos. \theta \frac{r}{L} \cos. \theta \right)$$

Weight of connecting rod = 30 pounds

$$M_C = \text{mass of connecting rod} = \frac{30}{32.2} = .953.$$

$$Z_C = 22.5 - 14.625 = 7.375$$

$$\frac{L - Z_C}{L} = \frac{7.375}{22.5} = .35$$

$$T_1 = \frac{(953. \times .35 \times .375 \times 56.25)}{25.959} \times \text{constant}$$

T₁ = .271 x constant, or by ratio of masses,

$$T_1 = \frac{30.5}{25} \times .35 \times F = .427 F.$$

Angle	F	T ₁ or .427 F
0°	- 29.60	- 12.63
30°	- 24.05	- 10.30
60°	- 9.97	- 4.25
90°	+ 4.98	+ 2.122
120°	+ 14.98	+ 6.40
150°	+ 19.25	+ 8.22
180°	+ 19.95	+ 8.53

$$T_2 = M_c \frac{Z_c}{L} W^2 r (\cos.\theta + \frac{r}{L} \cos. \theta)$$

$$\frac{Z_c}{L} = \frac{14.625}{22.5} = .65$$

$$T_2 = \frac{(.953 \times .793 \times .375 \times 56.25)}{25.96} \times \text{constant}$$

or

$$T_2 = (1.22 \times 65 \times F = .793 F$$

Angle	F	T2 or .793 F
0°	- 29.60	- 23.43
30°	- 24.05	- 19.10
60°	- 9.97	- 7.91
90°	+ 4.98	+ 3.95
120°	+ 14.98	+ 11.89
150°	+ 19.13	+ 15.20
180°	+ 19.95	+ 15.83

ANGULAR INERTIA FORCE OF CONNECTING ROD

The angular inertia force "A" of the connecting rod when rotating about the wrist pin may be determined from the equation:

$$A = M_c Z_c W^2 \frac{r}{L} \sin \theta$$

$$- M_c Z_c W^2 = \frac{.953 \times 4.625 \times 56.25}{12 \times 25.956}$$

$$x 4^2 = - 99.50.$$

Angle	$\sin \theta$	$1/5 \times \sin \theta$
0°	0	0
30	.5	.1
60	.866	.173
90	.1	.20
120	.866	.173
150	.5	.1
180	0	0

Angle	$1/5 \sin \theta$	Constant	A
0°	+ 0	- 99.50	0
30°	+ .1	"	- 9.94
60°	+ .173	"	-17.21
90°	+ .2	"	-19.85
120°	+ .173	"	-17.21
150°	+ .1	"	- 9.94
180°	0	"	0

" A_1 " and " A_2 " are two forces acting at the wrist pin and crank pin respectively, which together give the same result as " A " acting at the center of percussion.

$$A_1 = \frac{A(L - S)}{L}$$

$$t = \frac{s}{g} = \frac{.31416}{32.2} \quad S = 3.26 \quad t^2$$

10 swings in 7 seconds .7 sec. = time of one swing. .7 x 2 = 1.4 sec. = time of 1 period of connecting rod.

$$S_1 = 3.26 \times t^2 = 3.26 \times .7^2 = 1.597.$$

$$S = \frac{D^1}{D} (S_1 - D) + D. \quad D = 14.625" \quad D_1 = 14.625"$$

$$S = 1 (1.597 - 1.22) + 1.22 = 1.597 \times 12 = 19.17"$$

$$A_1 = A \frac{(22.5 - 19.17)}{22.5} = .148 A$$

Angle	Constant	A	A ₁
0°	.148	0	- 0
30°	"	- 9.94	- 1.47
60°	"	-17.21	- 2.55
90°	"	-19.85	- 2.94
120°	"	-17.21	- 2.55
150°	"	- 9.94	- 1.47
180°	"	- 0	0

$$A_2 = A \times \frac{S}{L} = \frac{19.17}{22.5} = .853 A.$$

Angle	Constant	A	A ₂
0°	.853	0	- 0
30°	"	- 9.94	- 8.47
60°	"	-17.21	-14.66
90°	"	-19.85	-16.91
120°	"	-17.21	-14.66
150°	"	- 9.94	- 8.47
180°	"	- 0	0

CENTRIFUGAL FORCE OF THE CONNECTING ROD

The centrifugal force "C" of the connecting rod acts at the center of gravity, is always

radial with the rod, and is outward in direction.

"C" can be determined from the equation

$$C = M_c W^2 Z_c \left(\frac{r}{L} \cos \theta \right)^2.$$

$$M_c Z_c W^2 = 4 \cdot 2 \times \frac{.953 \times 56.25 \times 14.625}{12 \times 25.96} = 99.50$$

Angle	$\cos \theta$	$1/5 \cos \theta^2$
0°	1	.04
30	.866	.0299
60	.5	.01
90	0	0
120	.5	-.01
150	.866	-.0299
180	1	-.04

Angle	$1/5 \cos \theta^2$	Constant	C
0°	.04	- 99.50	+3.98
30	.0299	"	+2.97
60	.01	"	+ .99
90	0	"	0
120	-.01	"	+ .99
150	-.0299	"	+2.97
180	-.04	"	+3.98

WEIGHT OF THE CONNECTING ROD

The influence of the weight of the connecting rod is slight and is often neglected.

$$G_1 = \frac{G (L - Z_c)}{L}$$

$$G_1 = \frac{22.5 - 14.625}{22.5} \times 30.5 = .412$$
$$\frac{22.5}{25.965}$$

$$G_2 = G \left(\frac{Z_c}{L} \right)$$

$$G_2 = \frac{14.625 \times 30.5}{22.5 \times 25.96} = .764$$

Since all the necessary forces have been calculated the resultant effect of the explosion pressure and inertia forces on the rotative effort of the crank pin, and on the shaking forces at the center of the main bearing can be determined.

The forces at the wrist pin are S_1 F_1 T_1 A_1 and G_1 resulting in a pressure on the guides and a thrust along the rod. This thrust along the rod combines with forces T_2 , A_2 , G_2 , and C_1 to produce (1) rotation of the crank shaft

(2) pressure on the main bearings

(3) An unbalanced force tending to shake the engine on its foundation.

These various resultants have been determined graphically in plates #1 and #2.

COUNTER BALANCE

The counter balance curve illustrates the principle of reducing the effect of the shaking forces by placing a counter balancing weight in the crank disc of the engine opposite to the crank pin. The centrifugal force of this weight is uniform for a constant speed and is always radial.

TURNING EFFORT

The force T E found on plate I in the force polygons acts at right angles to the crank and represents the turning effort in pounds per sq. in. of piston area. While the piston makes a stroke equal to twice the crank length, or $2r$, the crank passes a distance r . Hence, in Fig. I, plate II, the turning effort at the various crank positions is laid off from the datum line equal to $4r$ (2 revolutions per cycle

in a 4 cycle engine) at the point corresponding to the crank pin position, in a circular path. This process results in a curve and the area between it and the datum line represents an amount of work in foot pounds (equal to that of the indicator card).

From the form of the turning effort curve it follows that if the load is constant the speed of the engine must fluctuate during each stroke. The mean ordinate represents the uniform crank effort which would do the same work per revolution as the actual effort. During part of the stroke, the flywheel gains speed and stores up kinetic energy which is restored during the remainder of the stroke.

CALCULATION OF VELOCITY AND DISPLACEMENT CURVES

$F = S$ (Scale of spring) $\times P$ (ordinate in inches taken from mean ordinate of turning effort curve).

$A =$ area of piston $= 25.959$ sq. inches.

$$dV = r dt = \frac{FR}{MK^2} \quad K = \frac{15}{12}$$

$$r = \frac{(4.5)}{12} \quad K^2 = \left(\frac{15}{12}\right)^2$$

$$dV = \frac{FR}{MK^2} r dt$$

$$Nt \text{ of 2 flywheels} = 2 \times 204 = 408$$

$$dt = \frac{60}{450 \times 36} = \frac{1}{270} = .00371.$$

$$dV = \frac{P \times 20 \times 25.959 \times 4.5^2 \times 12^2 \times 32.2 \times .00371}{12^2 \times 15.5^2 \times 408}$$

$$= dV = .0127 P F T.$$

"DV" is then the velocity imparted to the flywheel, measured on the crank pin circle, during each interval by the force corresponding to P . The dV will then represent the total velocity imparted to the crank pin, measured from any given starting point.

The crank pin velocity curve is found by plotting dV 's on a datum line, 4 r inches long,

divided into 36 equal parts. Each of the 36 equal parts correspond to those of the turning effort curve. The resulting curve represents the variation in velocity of the crank pin throughout 2 revolutions. By means of a planimeter the mean ordinate of this curve was found, and drawn giving a line of average velocity of crank pin.

After getting the crank pin velocity curve, the crank pin displacement curve is found in a similar manner.

$$ds = dv_s dt = .00371 dv_s.$$

Since ds is the increment of space passed over in each interval, it is evident that the $ds =$ total distance passed over by the crank pin during any number of intervals. This summation was plotted for every 10° and the resulting curve of Fig. 3, plate II obtained. The mean ordinate of this curve represents the displacement of the crank pin, or the distance in feet through which the crank would pass in unit time if its motion were uniform.

TABLE I

TABULATION OF FORCES

0°	F	T	T ₁	T ₂	A	A ₁	A ₂	C	G ₁	G ₂
0	-29.60	-	36.06	-12.63	-	23.43	-	0	-	0
30	-24.05	-	29.40	-10.50	-	19.10	-	9.94	-	1.47
60	-9.97	-	12.16	-4.25	-	7.91	-	17.21	-	2.55
90	+ 4.98	+	6.072	+ 2.122	+	3.95	-	19.85	-	2.94
120	+14.98	+	18.29	+ 6.40	+	11.89	-	17.21	-	2.55
150	+19.13	+	23.42	+ 8.22	+	15.20	-	9.94	-	1.47
180	+19.95	+	24.36	+ 8.53	+	15.83	-	0	-	0
210	+19.13	+	23.42	+ 8.22	+	15.20	+	9.94	+	1.47
240	+14.98	+	18.29	+ 6.40	+	11.89	+	17.21	+	2.55
270	+ 4.98	+	6.072	+ 2.122	+	3.95	+ 19.85	+	2.94	+ 16.91
300	- 9.97	-	12.16	- 4.25	-	7.91	+ 17.21	+	2.55	+ 14.66
330	-24.05	-	29.40	-10.30	-	19.10	+ 9.94	+ 1.47	+ 2.47	2.97
360-	-29.60	-	36.06	-12.63	-	23.43	+	0	+	0
720										

TABLE 2

VELOCITY

Nº	θ°	P"	Px. 0127 or dv = ft.	dv ft.
1	5	.04	.0005	.0005
2	15	3.00	.0381	.0386
3	25	7.45	.0945	.1331
4	35	7.33	.0930	.2261
5	45	6.57	.0834	.3095
6	55	5.50	.0698	.3793
7	65	4.44	.0564	.4357
8	75	3.70	.0470	.4827
9	85	3.23	.0410	.5237
10	95	2.95	.0374	.5611
11	105	2.75	.0348	.5959
12	115	2.48	.0314	.6273
13	125	2.17	.0276	.6549
14	135	1.83	.0232	.6781
15	145	1.32	.0162	.6943
16	155	.53	.0067	.7010
17	165	0	0	.7010
18	175	-.39	-.0049	.6961
19	185	-.68	-.0085	.6876
20	195	-.9	-.0115	.6761
21	205	-.1.12	-.0142	.6619
22	215	-.1.26	-.0160	.6459
23	225	-.1.38	-.0175	.6284
24	235	-.1.46	-.0185	.6099
25	245	-.1.46	-.0185	.5914
26	255	-.1.34	-.0169	.5755
27	265	-.1.14	-.0145	.5610
28	275	-.9	-.0114	.5496
29	285	-.53	-.0067	.5429
30	295	-.08	-.00111	.5418
31	305	+.22	-.0028	.5446
32	315	+.39	-.00445	.5480
33	325	+.40	-.0057	.5537
34	335	+.35	-.00445	.5581
35	345	+.2	-.00254	.5606
36	355	-.18	-.00228	.5583
37	365	-.83	-.0105	.5478
38	375	-.1.23	-.0156	.5322
39	385	-.1.44	-.0183	.5139
40	395	-.1.53	-.0194	.4945

TABLE 2 (continued)

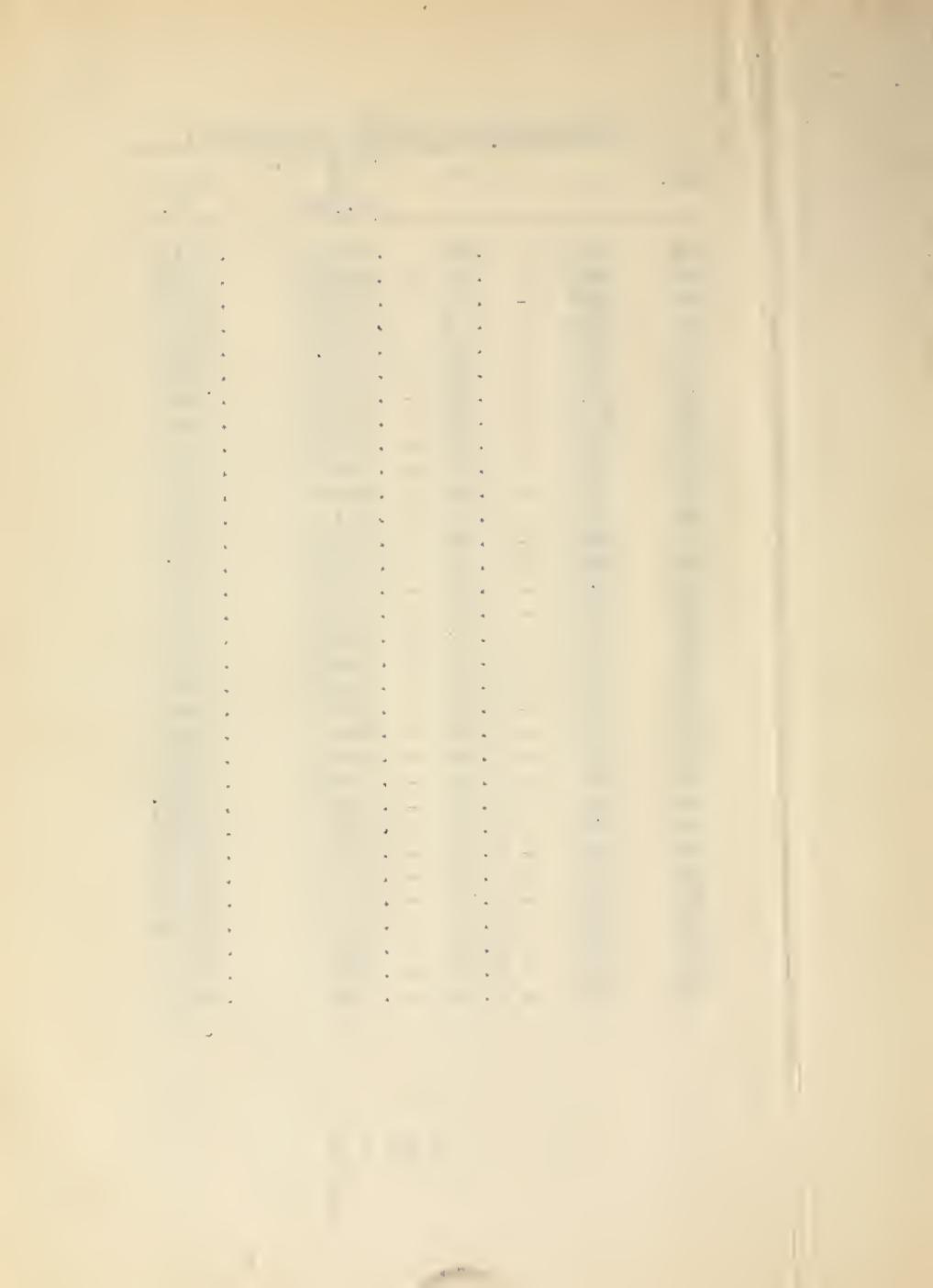
VELOCITY					
N ^o	θ ^o	P"	Px.0127 or dv = ft.	dv =	ft.
41	405	- 1.53	- .0194	.4751	
42	415	- 1.44	- .0183	.4568	
43	425	- 1.34	- .0170	.4398	
44	435	- 1.24	- .0157	.4241	
45	445	- 1.18	- .0156	.4091	
46	455	- 1.06	- .0135	.3956	
47	465	- .19	- .0024	.3932	
48	475	+ .23	- .0029	.3961	
49	485	.28	- .0036	.3997	
50	495	.20	- .0025	.4022	
51	505	.08	- .0011	.4033	
52	515	- .07	- .0009	.4024	
53	525	- .2	- .00254	.3999	
54	535	- .38	- .00482	.3951	
55	545	- .62	- .0078	.3874	
56	555	- .90	- .0114	.3759	
57	565	- 1.25	- .0158	.3601	
58	575	- 1.42	- .0180	.3421	
59	585	- 1.47	- .0185	.3236	
60	595	- 1.46	- .0185	.3051	
61	605	- 1.42	- .0179	.2872	
62	615	- 1.42	- .0179	.2693	
63	625	- 1.46	- .0185	.2508	
64	635	- 1.46	- .0185	.2323	
65	645	- 1.42	- .0179	.2144	
66	655	- 1.32	- .0165	.1979	
67	665	- 1.35	- .0169	.1810	
68	675	- 3.40	- .0425	.1385	
69	685	- 4.40	- .0550	.0835	
70	695	- 4.29	- .0535	.0300	
71	705	- 3.07	- .0380	-.0080	
72	715	- 1.29	- .0161	-.024	

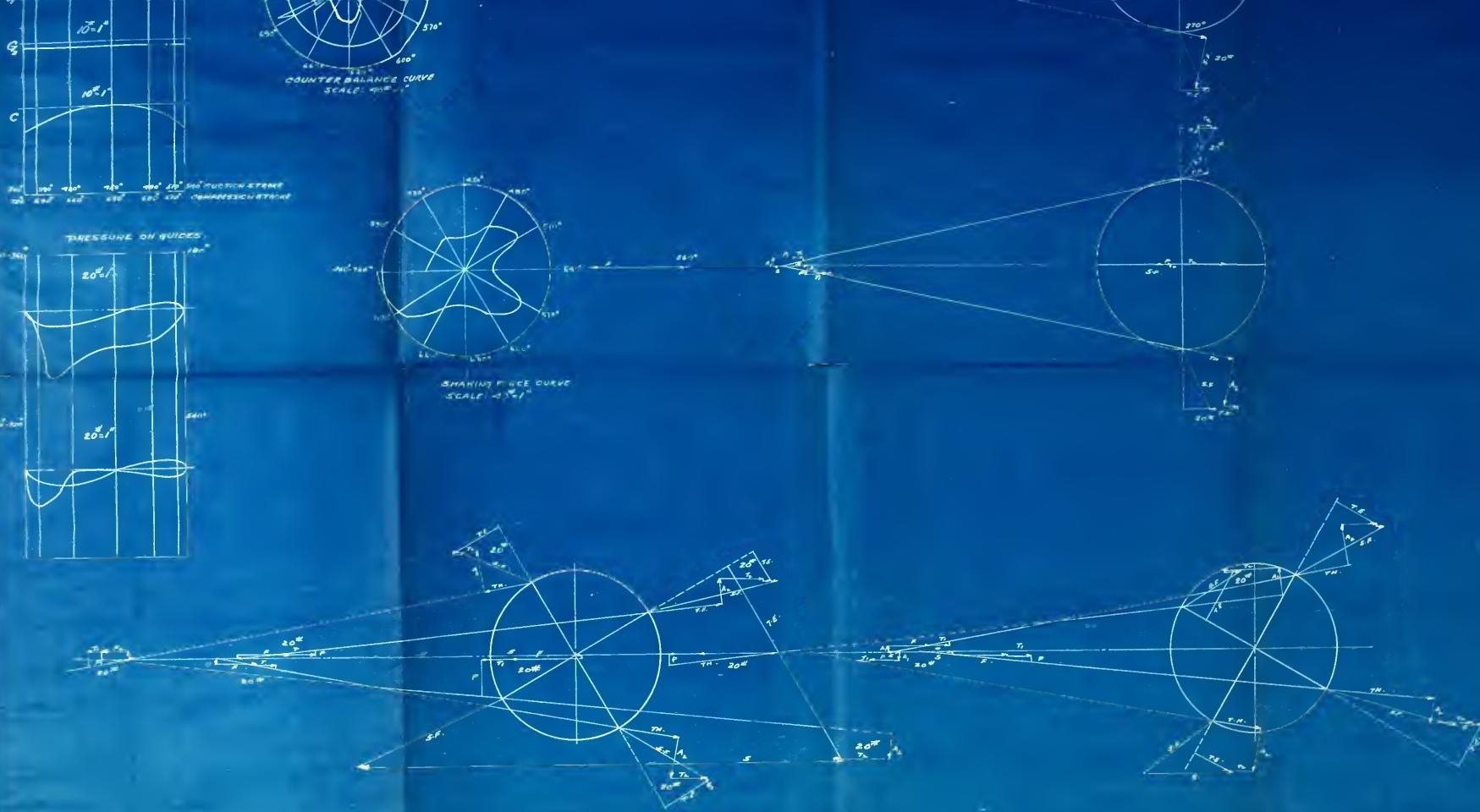
DISPLACEMENT TABLE

N ^o	θ	P"	Px.0037		ds = ft.
			or	dv _s = ft.	
1	10	- 4.02	- .0148	- .0148	
2	20	- 3.40	- .0126	- .0274	
3	30	- 2.35	- .0078	- .03526	
4	40	- 1.5	- .0056	- .04082	
5	50	- .64	- .0024	- .04319	
6	60	- .08	- .00029	- .04348	
7	70	+ .43	+ .00159	- .04189	
8	80	+ .80	+ .00296	- .03893	
9	90	+ 1.21	+ .0045	- .03443	
10	100	+ 1.55	+ .00575	- .02868	
11	110	+ 1.88	+ .00696	- .02172	
12	120	+ 2.22	+ .00816	- .01356	
13	130	+ 2.48	+ .0092	- .0043	
14	140	+ 2.65	+ .00985	+ .0054	
15	150	+ 2.77	+ .0103	+ .01519	
16	160	+ 2.77	+ .0103	+ .0261	
17	170	+ 2.70	+ .0100	+ .0361	
18	180	+ 2.60	+ .00965	+ .0451	
19	190	+ 2.56	+ .0095	+ .0552	
20	200	+ 2.30	+ .0085	+ .0636	
21	210	+ 2.16	+ .008	+ .0716	
22	220	+ 1.97	+ .0073	+ .0789	
23	230	+ 1.79	+ .0066	+ .0855	
24	240	+ 1.62	+ .0060	+ .0915	
25	250	+ 1.47	+ .0055	+ .0970	
26	260	+ 1.35	+ .00493	+ .1019	
27	270	1.24	.0046	.1065	
28	280	1.20	.0045	.1109	
29	290	1.24	.0046	.1155	
30	300	1.27	.0047	.1202	
31	310	1.32	.0049	.1252	
32	320	1.36	.0050	.1302	
33	330	1.39	.0051	.1353	
34	340	1.37	.0051	.1404	
35	350	1.30	.0048	.1452	
36	360	1.19	.0044	.1496	
37	370	1.02	.00379	.1534	
38	380	.84	.00312	.1565	
39	390	.63	.00234	.1588	
40	400	.42	.00156	.1604	

DISPLACEMENT TABLE (continued)

No.	θ	P"	ds = ft.		ds = ft.
			or	Px.0037	
41	410	.26	.00096		.1614
42	420	+.10	+.000371		.1617
43	430	-.06	-.00022		.16150
44	440	-.2	-.00074		.1608
45	450	-.28	-.00104		.1587
46	460	-.30	-.00111		.1576
47	470	-.28	-.00104		.1557
48	480	-.22	-.00082		.1548
49	490	-.20	-.00074		.1541
50	500	-.21	-.00077		.1533
51	510	-.24	-.00089		.1524
52	520	-.27	-.001		.1514
53	530	-.32	-.00118		.1502
54	540	-.42	-.00156		.1487
55	550	-.54	-.0020		.1467
56	560	-.71	-.00263		.1440
57	570	-.9	-.00334		.1407
58	580	1.09	-.00404		.13669
59	590	1.26	-.00467		.1320
60	600	1.45	-.00537		.12665
61	610	1.65	-.0061		.12055
62	620	1.84	-.0068		.11373
63	630	2.04	-.00756		.10617
64	640	2.21	-.0082		.09697
65	650	2.39	-.0088		.08810
66	660	2.65	-.0098		.07826
67	670	3.17	-.0117		.06656
68	680	3.72	-.0138		.05276
69	690	4.09	-.0152		.03756
70	700	4.20	-.0155		.0220
71	710	4.23	-.0157		.0063
72	720	4.23	-.0157	-	.0094





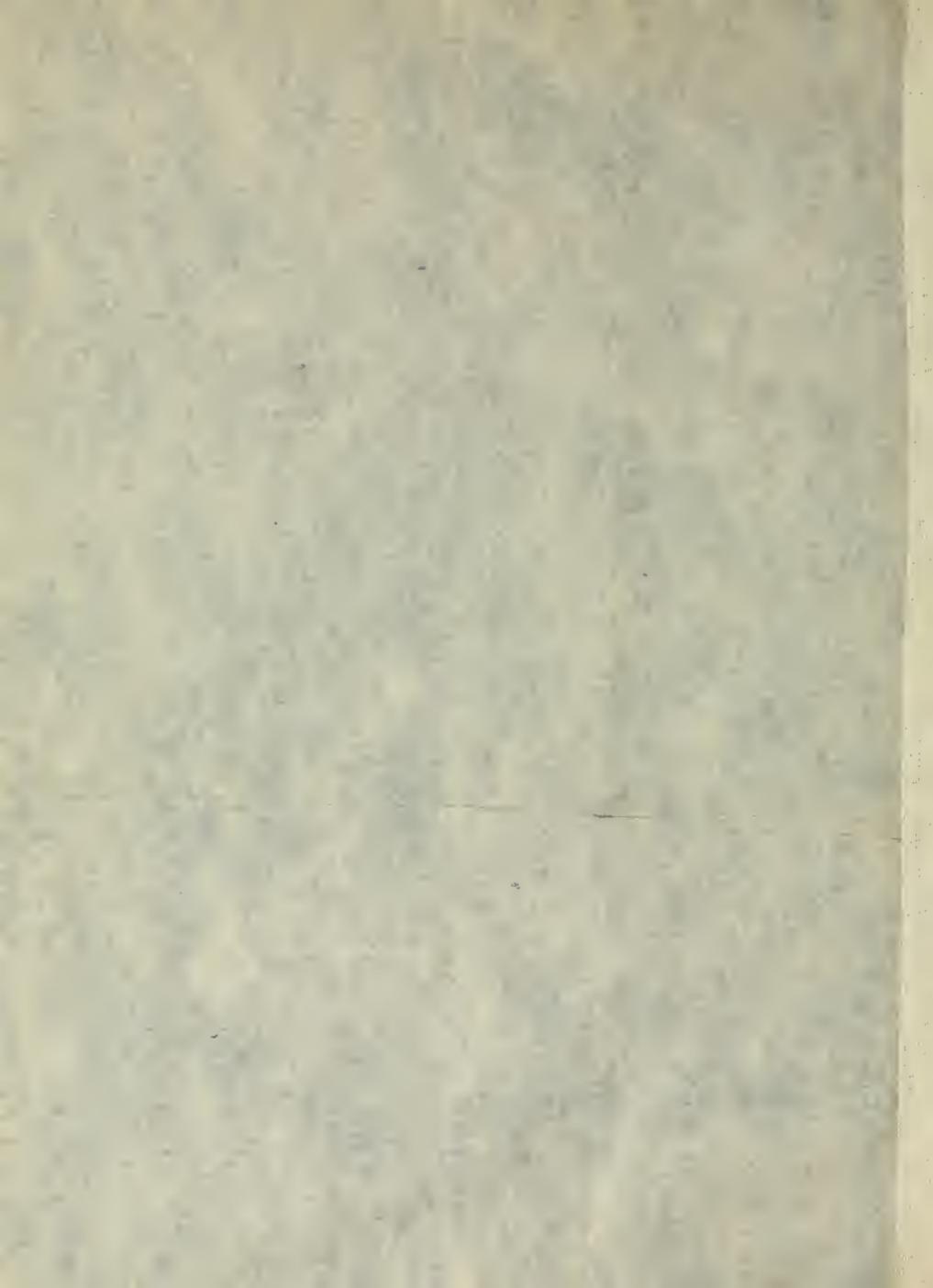




PLATE 2



SHAKING FLAMES

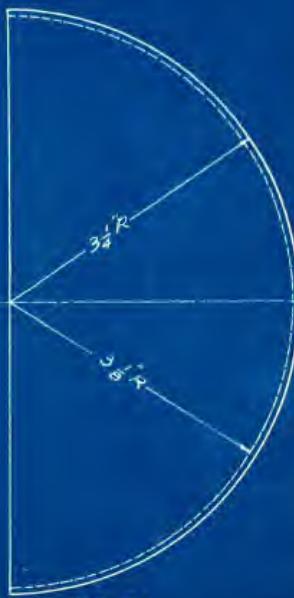
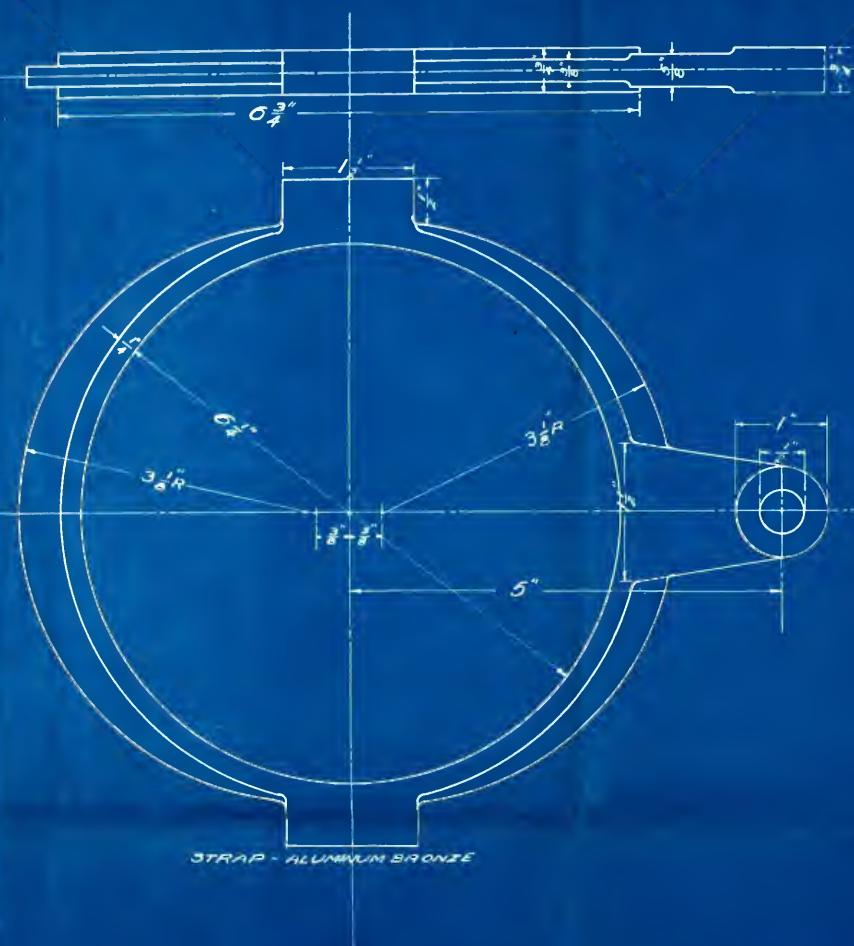
6 H.P. HYD ENGINE

450 R.P.M.

5000 K.C.P.H. THERM

10000 K.C.P.H. THERM



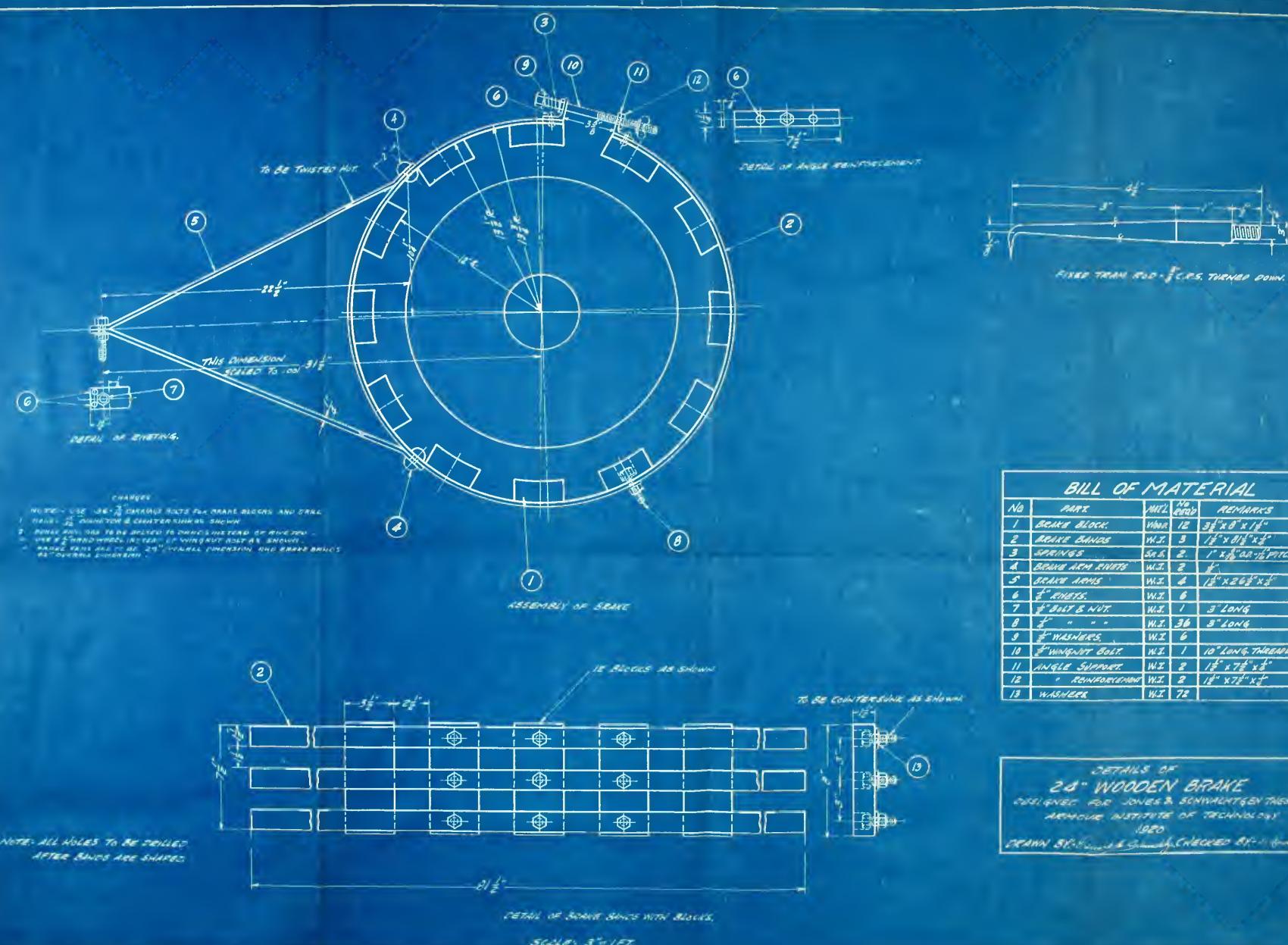


SHEAVE - C.

STRAP - ALUMINUM BRONZE

DESIGN OF
ECCENTRIC
FOR RECIPROCATING MOTION ON
AVIATION ENGINE
MAILED 20. 20. 1940. THESIS
LE JONES
AIAE





BILL OF MATERIAL			
NO.	PART	SIZE	REMARKS
1	BRAKE BLOCK	W.H. 12	3 1/2" x 10" x 1 1/2"
2	BRAKE BANDS	W.I. 3	1 1/2" x 10" x 3"
3	SPRINGS	S.A.S. 2	1" x 10" OR 10" PITCH
4	BRAKE ARM RIVETS	W.I. 2	1/8"
5	BRAKE ARMS	W.I. 4	1 1/2" x 26 3/8" x 3"
6	1/8" RIVETS	W.I. 6	
7	3/8" BOLT & NUT	W.I. 1	3" LONG
8	3/8" " "	W.I. 36	3" LONG
9	3/8" WASHERS	W.I. 6	
10	3/8" WING-NUT BOLT	W.I. 1	10" LONG THREADED
11	ANGLE SPANNER	W.I. 2	1 1/2" x 7 3/8" x 3"
12	1" REINFORCEMENT	W.I. 2	1 1/4" x 7 3/8" x 3"
13	WASHER	W.I. 72	

**DETAILS OF
20" WOODEN BRAKE**
DESIGNED FOR JONES & SCHWARTZENGEN THERES
ARIZONA INSTITUTE OF TECHNOLOGY
1920
DRAWN BY [unclear] CHECKED BY [unclear]

